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Review of Two-phase Electronics Cooling for Army Vehicle Applications

by Darin Sharar, Nicholas R. Jankowski, and Brian Morgan

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14. ABSTRACT
Implementation of many Army vehicle electronics technologies is becoming increasingly dependant on the ability to dissipate large amounts of waste heat. In the past, these technologies have been primarily cooled with active and passive air or single-phase heat sinks. While single-phase active and passive cooling methods are well-characterized, making system implementation predictable and dependable, these schemes are not providing the cooling performance necessary for future high-heat-flux electronic systems. By comparison, two-phase cooling has demonstrated the ability to provide net benefits in heat dissipation, size, weight, and overall power consumption necessary for future electronic systems. However, unlike single-phase cooling, two-phase cooling performance is difficult to estimate, making system integration unpredictable and often unreliable. Therefore, two-phase research and system integration have become major focal points for academic and professional organizations, alike, aimed towards resolving the uncertainties associated with two-phase cooling and developing the next generation of high-heat dissipation techniques. This report reviews the major two-phase cooling approaches in context of Army vehicular applications.

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1. Introduction

1.1 Army Electronics Cooling Needs

The increasing development of electric platforms has been a growing trend across the separate branches of the U.S. Department of Defense (*1–5*). The platform proponents hope that replacing many of the traditional mechanical power and control systems with electronic equivalents will enable significant improvements by reducing weight, lowering fuel usage, increasing design flexibility, and enhancing overall functionality (*6*). In addition to work in the commercial sector, this trend is apparent in "drive by wire" and "fly by wire" system designs (*7, 8*). Specific to U.S. Army platform development, the past few iterations of Army vehicle programs have focused on increasing the use of electrical systems throughout the vehicle for many of the same reasons (*6, 8–11*). Some of these programs have focused on new fully hybrid vehicles, while others have looked at improvements that can be made to legacy systems (*8*). Because the disparate electrical systems common to these programs need to draw power from a platform's common electrical bus, they all require improved power conversion electronics to meet operational specifications.

U.S. Army Tank Automotive Research, Design and Engineering Center (TARDEC) and the U.S. Army Research Laboratory (ARL) have been actively investigating hybrid electric vehicle architectures, power electronic subsystems, and components in support of this development strategy (*6*, *12*). Aligned efforts aimed towards increasing the component power while simultaneously decreasing component size have led to improvements in power density. At the same time, however, increasing power density directly exacerbates the task of managing waste heat while staying within acceptable temperature limits for these components (*9*); new generation electronic systems are producing multiple kilowatts of waste heat with projected future device level heat fluxes upwards of 1000 W/cm^2 (*13*).

Most vehicle designs have relied on air-cooled heat sinks for low-heat-flux electronics and an antifreeze-based automotive liquid coolant loop to manage the larger waste heat from engine compartment components (*11*, *14*). However, new power-dense electronic systems are further increasing waste heat and presenting great challenges to the capabilities of conventional air and single-phase liquid cooling systems (*2*). Considering strictly air cooling, the effect of higher heat flux electronics is larger, heavier, costlier heat sinks and fans to compensate for insufficient convective performance. The effect is equally dramatic with single-phase liquid cooling, with higher heat fluxes requiring larger coolant flow rates to sufficiently cool the system devices (*15*). These large flowrates and subsequent pumping powers result in increasingly bulky, heavy systems that consume more fuel (*10*). Thus, there is a drive to develop improved cooling components that are smaller and lighter, and have increased convective performance relative to conventional liquid cold plates.

Over the past several decades, the academic and industrial research communities have explored many options for improving liquid cold plate convective performance. These options have included implementing various micro- and mini-channel designs to increase heat transfer surface area, as well as using alternative fluid delivery methods in the forms of both macro- and micro-scale jets and sprays (*16–18*). No matter the particular heat sink configuration, the automotive cooling options that have been pursued beyond a laboratory environment have almost exclusively been single-phase liquid flow designs (*7*). Extensive single-phase research and design iterations have, until recently, provided sufficient thermal performance for cooling Army vehicles.

With ever-increasing electronic heat fluxes and mounting single-phase thermal management issues, cooling schemes using liquid-vapor phase change (hereafter referred to as “two-phase cooling” in this report) have been examined as a practical and cost-conscious next-steps beyond single-phase cooling. In fact, the Department of Defense (DoD) and the National Aeronautics and Space Administration (NASA) have already distinguished two-phase flow technologies as a favorable solution to meeting the strict demands for some emerging cooling requirements (*3*, *4*). Despite these endorsements, two-phase cooling schemes have traditionally not been used due to technical risks related to flow, pressure, and critical thermal failure. This report will review both the potential advantages and disadvantages of two-phase cooling technology within the context of Army vehicle power system thermal management, and propose paths for future research.

1.2 Army Vehicle Platform Considerations

We examined the Army vehicle platform and the constraints it imposes upon a particular cooling solution. As previously mentioned, most vehicles have a single-liquid coolant loop for the removal of waste heat from the engine and other “under hood” components. This has historically been an antifreeze loop consisting of a 50/50 ethylene glycol and water (EGW) solution (*14*, *19*), though there has been a recent commercial movement toward the use of lower toxicity propylene glycol and water (PGW) solutions (*20*). This loop is driven by a single-coolant pump, and after acquiring heat from various heat sources will pass through a liquid-air heat exchanger (typically a fan-cooled radiator) to reject heat to the environment.

The temperature at which the coolant is delivered to the electronic components is highly dependent on operating condition. Current Army vehicle requirements call out inlet temperatures around 65 °C, with future objectives of 80–100 °C (*9*) because higher coolant temperatures increase efficiency and improve power and weight savings on the radiator side of the cooling loop (*21*). Most silicon power electronics have a full-power operating temperature limit of about 125 °C (*22*), and the boiling temperatures for EGW and PGW at ambient pressures are 107 °C and 106 °C, respectively. These high inlet temperatures limit the electronic device temperature increase to only 25–45 °C above inlet temperature, reducing electronic cooling potential and requiring exceptional cooling performance to dissipate high-heat-fluxes and prevent overheating.

In addition to low allowable temperature margins, the EGW coolant, itself, limits the electronics thermal budget. EGW is used mainly to provide a wider liquid temperature range for the fluid than a water system would provide (–40 °C–107 °C compared to 0 °C–100 °C at ambient pressure, respectively). However, past research on single-phase cooling has shown that EGW and PGW mixtures can have a 28% to 42% reduction, respectively, in cooling performance compared to a pure water microchannel system (*23*), mainly due to increased viscosity and decreased specific heat of the EGW and PGW mixtures compared to pure water. Furthermore, typical cooling systems are not even pure 50/50 EGW. Additives that affect fluid properties—borax, sodium Mercaptobenzothiazole (sodium MBT), trisodium phosphate, and antifoaming agents—are often present. It is unclear exactly what effect these additives have on thermal performance or the function of advanced heat exchangers (*19*).

The single coolant loop architecture also restricts the potential performance of the electronic cooling components. Because the engine is usually the primary cooling concern for this loop, changes to loop parameters, such as fluid type, flowrate, and available pressure, generally cannot be made even if electronics cooling performance could be improved. The loop is generally high flowrate with low pressure drop available for electronics cooling (typically 2–5psi), which complicates or even prevents the use of many cooling solutions that often have driving pressure requirements on the order of 10–100 psi (*16, 24, 25*). While these designs may reduce total pumping power, integration to current single loop architectures would require flow conversion.

This single loop architecture is also not designed to handle combined gas and liquid flow, as would occur in boiling heat transfer. Consequently, vapor buildup in any portion of the system could result in coolant venting or reduced coolant flow through the radiator and premature thermal failure due to subsequent reduced heat transfer. Therefore, a margin of safety is required to maintain the system well within single-phase conditions, which can increase flowrate, pumping power, and radiator requirements. Additionally, the specific boiling behavior of EGW and PGW (termed binary fluids) is typically suppressed compared to the onset conditions for boiling of pure components (*26*). The resulting suppression, whose severity is dependent on the thermodynamic properties of the specific mixture, presents an additional resistance to heat transfer.

Finally, while the engine coolant loop is a sealed system, it is user-serviced and subject to contamination and particulate accumulation (*19*). From a design perspective, contamination either requires a system with large enough fluid lines and cooling geometries to provide immunity to clogging (*27*), or stringent filtering requirements to remove contamination from the system. Both of these scenarios have negative consequences; typically larger geometries in cold plates result in reduced thermal performance, and rigorous filtering may cause large system pressure drops and reduced flowrates.

Because of the limitations of a single coolant loop, some consideration has been given to a secondary cooling loop that either exchanges heat with the primary coolant loop or with outside

air directly. This approach is most readily apparent in the cabin cooling or air conditioning systems present in almost all modern vehicles. Beyond this, other platforms have examined the use of secondary loops specifically designed for cooling non-engine electronic heat loads (*3, 7*). Four potential benefits lie in the ability to design a secondary cooling loop optimized for the specific heat loads in the system:

1. Flexibility of working fluid, including refrigerants in some cases. Table 1 in appendix A shows typical coolant fluids with compiled properties.
2. Possibility to implement a vapor compression refrigeration cycle to deliver coolant to the cold plates below ambient temperature.
3. Pump autonomy to optimize system pressure, flow, and temperature to match specific heat loads.
4. Potential for hermetically sealed systems to reduce contamination and clogging/fouling potential.

There are definite disadvantages involved with implementing a secondary loop; the most obvious of these are the additional size, weight, and power (SWAP) required for the extra loop components such as pumps, hoses/tubing, and heat exchangers/condensers. While the possible drawbacks of such a two-loop system may be significant, the overall potential for cooling performance improvement may be able to offset those negative elements. Implementation of a secondary loop may add system complexity but enables the use of thermal techniques, like microchannels and two-phase cooling, that would be logistically difficult with a primary engine coolant loop. Two-phase cooling has demonstrated a huge potential that could tip balance in favor of a secondary loop.

The remainder of this report will discuss potential advantages of these two-phase systems, briefly describe three competing two-phase cooling schemes, and describe one scheme in-depth (two-phase microchannel coolers) that has the most potential to cool vehicular applications with current Army specifications in mind. In addition, we will discuss impediments to two-phase microchannel cooling techniques, examining steady-state research and introducing a relatively unmitigated obstacle, two-phase microchannel transient behavior.

2. Two-Phase Cooling Background

Two-phase cooling refers to intentionally using a cooling fluid in a manner where at least a portion of the fluid is transformed into vapor upon heating, thereby resulting in a gas/liquid mixture in a portion of the cooling loop. The boiling event generally occurs when the temperature of the heat acquisition surface exceeds the liquid's saturation temperature, or boiling point. At this temperature, the vapor pressure of the liquid is equal to the pressure of the

environmental surroundings, thus allowing vapor bubbles to form at the solid-liquid interface, grow, and eventually detach from the surface. After this point, multi-phase flow exists in the system until the vapor is either separated from the liquid flow or is cooled to the point of condensing back into a liquid; schematically, the components typically used (the filter and vapor/liquid separator are optional) to achieve a two-phase loop are presented in figure 1. A liquid may be induced to boil over a range of temperatures, depending on the pressure of the environment, where increasing system pressure will increase the nominal boiling temperature of the fluid. This provides an additional benefit for a system using two-phase cooling, whereby the boiling and condensation temperatures of enclosed coolant loops may be tailored by controlling system pressurization.

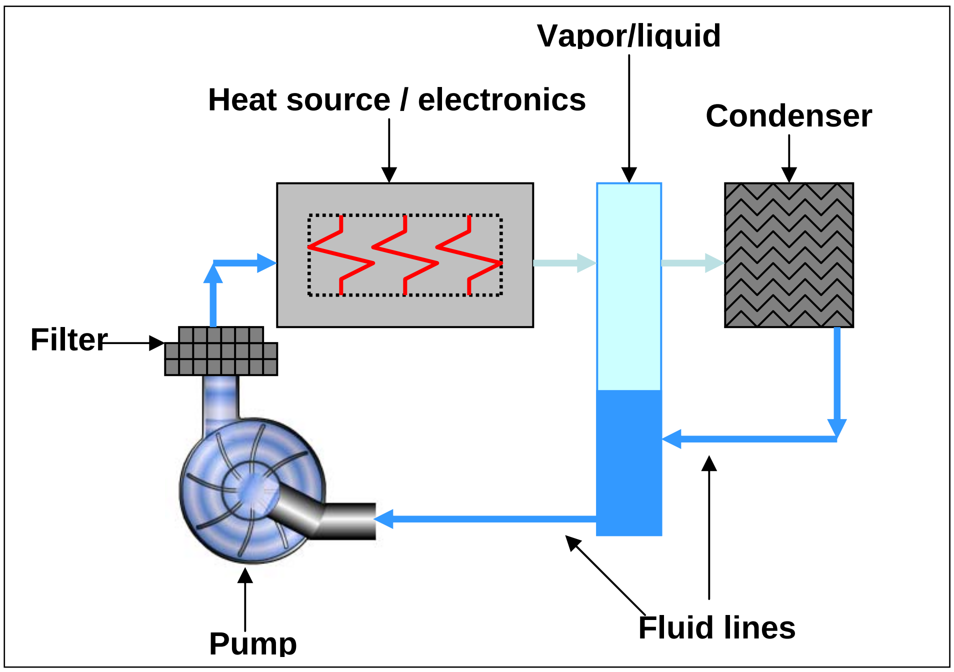

Figure 1. Primary components of a two-phase cooling loop.

A two-phase system has several potential advantages over a standard single-phase liquid cooling system. First, a fluid's latent heat of vaporization has a fundamental limit that can be two orders of magnitude larger than the specific sensible heat of single-phase liquid cooling (*28*). Thus, the boiling effect provides the possibility of increased heat absorption per unit volume of fluid and higher heat acquisition effectiveness (i.e., the amount of heat absorbed by a unit of flow relative to its maximum theoretical capacity). This latent heat benefit is coupled with the improved convection due to buoyancy-driven bubble formation, multi-phase turbulation, and mixing that takes place within the heat transfer region. As shown in figure 2, the resultant heat transfer coefficients from two-phase flow can be an order of magnitude greater than equivalent single-phase forced or passive (natural) convection (*28*). A larger heat transfer coefficient translates directly into a larger cooling capability for the system. A typical rule of thumb for two-phase liquid cooling systems is that the heat of vaporization and convective benefits of boiling can

yield a two-to-four-times increase in heat removal capability (*14*). This performance projection is currently limited by safety factors and system derating associated with uncertainties of two-phase cooling; by developing a better understanding of the process, engineers may be able to take full advantage of the vast convection and latent heat improvements.

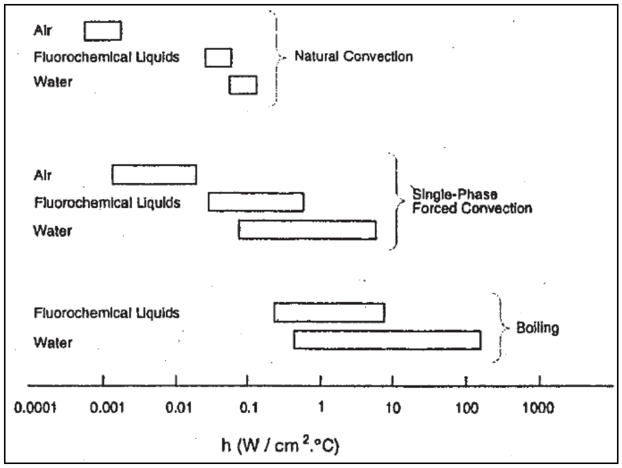

Figure 2. Heat transfer coefficients possible with natural convection, single-phase liquid forced convection and boiling for various fluids; from (*28*).

In addition to improved cooling capacity, two-phase cooling also has the potential benefit of providing a relatively isothermal cooling surface (possibly less than 1 °C) (*14, 29, 30*). Whereas sensible heating of a single-phase coolant creates a downstream temperature rise, inducing a lateral temperature gradient along the cooled surface, two-phase coolers use latent heat absorption for energy transfer, which occurs at nearly isothermal conditions. Because excessive temperature and cooling non-uniformity over the electronic device area can induce thermal stresses and potential failure, isothermal cold plates can greatly increase the functional reliability of attached components (*5*).

Weight, volume, and cost reduction are additional potential benefits of two-phase cooling techniques. Due to the increased effectiveness of two-phase cooling, lower flowrates can be used to deliver cooling performance similar to high flowrate single-phase systems. The potential value of lower system flowrates includes: smaller pumps with lower pumping power requirements; smaller onboard fluid volume; smaller fluid reservoirs; smaller tubing; and a reduction in related packaging weight and volume.

During the past several decades of two-phase cooling research, three primary cooling schemes have gained the most attention: spray cooling, jet-impingement, and microchannel cooling (*28*). It is worth noting that each of these techniques can be used for single-phase or two-phase operation. During startup conditions, even a two-phase system has to move through a liquid-only single-phase regime. Proper system design would permit successful transient passage through this regime to the steady two-phase design point. The following sections will discuss the

advantages and disadvantages of each of these techniques, mainly focusing on two-phase cooling characteristics.

2.1 Spray Cooling

In figure 3, the primary characteristics of a spray cooling configuration can be seen. Fluid is introduced to an orifice or nozzle offset from the heated surface. With appropriate driving pressure, the fluid leaving the orifice breaks up into individual droplets that impinge upon the hot surface. The fluid will form a film upon the surface, the thickness of which is dependent on the relative fluid removal rate. With sufficient heating of the hot surface, there will be bubble formation in the liquid film, and the system will be operating as a two-phase cooler.

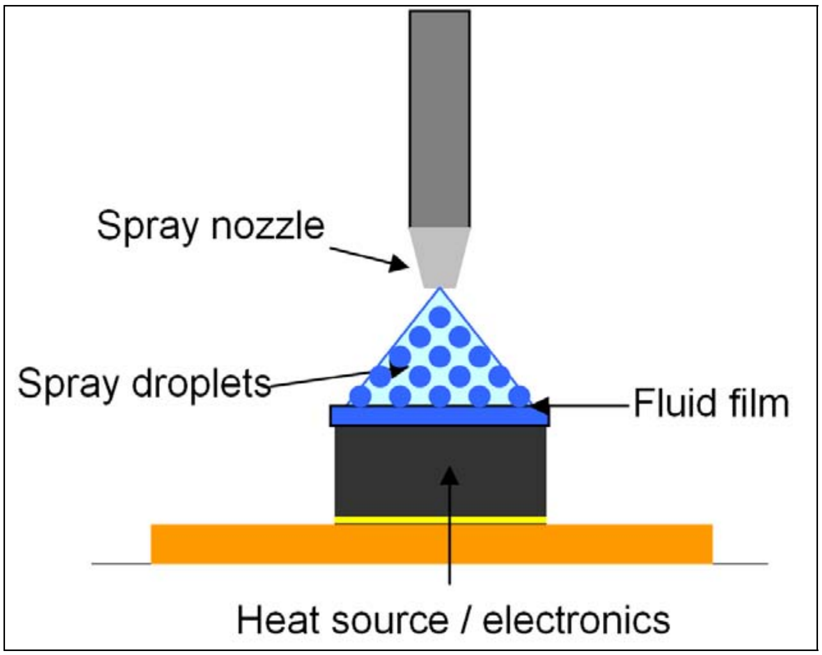

Figure 3. Primary components of a spray cooler.

Figure 4 shows qualitatively the relatively even cooling distribution that spray cooling can produce, primarily as a result of the uniformity of the spray pattern on the heated surface. Along with the other previously mentioned potential benefits of two-phase cooling, this can provide a uniform heat flux and temperature across the electronics. This fluid delivery method not only increases the spatial uniformity of heat removal but also delays liquid separation and eventual burn-out during vigorous boiling (*28*). As discussed in more detail in section 3.2, liquid separation refers to a vapor blanket present between the coolant fluid and heat source and is one condition leading to potential system overheating and failure. For the most part, spray coolers also provide low flow rate requirements due to the precise fluid-delivery and heat removal mechanism.

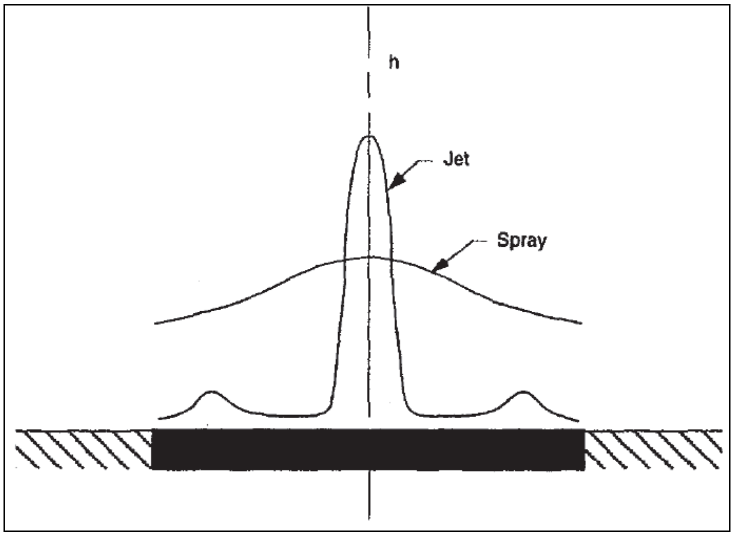

Figure 4. Qualitative comparison for heat transfer coefficients of spray coolers and jet impingement coolers; from (*28*).

Despite these advantages, spray cooling is considered excessively complex and unfeasible for many applications (*1*, *31*). Most spray cooling demonstrations have used standard commercial nozzles not necessarily designed or optimized for cooling applications. These nozzles generally require very large driving pressures (10–45 psi or even higher) to induce droplet breakup (*24*), and even supposedly identical nozzles can exhibit a definite lack of spray pattern repeatability and heat transfer performance (*18*). In addition, erosion, corrosion, and nozzle clogging are primary concerns for practical implementation that may require design concessions that degrade thermal performance (*18*).

The modeling and prediction of spray cooling performance also suffers from the sheer complexity of the fluidic and thermal behavior or the system. This complexity is due to the large number of parameters associated with spray cooling performance, many of which are hard to control or characterize in an experimental setup. In addition to parameters important for two-phase cooling, in general, such as fluid and heated surface properties, other spray parameters include nozzle type and shape, droplet size, volume flux, spray angle, nozzle-to-surface distance, inclination with respect to gravity, and spray overlap—all of which can significantly affect cooling performance (*32*–*46*).

Even though two-phase spray coolers have demonstrated drawbacks, research efforts have shown the potential to provide high-heat-flux electronics cooling. In a study by Lin and Ponnappan (*24*) using FC-87 and methanol, heat fluxes of 91.5 W/cm^2 and 490 W/cm^2, respectively, were demonstrated before overheating and physical burn-out of the electronics occurred. Mudawar et al. (*7*) developed a spray cooler as part of the U.S. Department of Energy's (DOE's) Power Electronics and Electric Machines program for developing hybrid-electric vehicles. A two-phase spray cooling module was implemented using HFE-7100 dielectric coolant fluid to safely dissipate 150-200W/cm^2 while maintaining the chip temperature below 125 °C. Similarly, Turek

et al. (*47*) developed a two-phase spray cooling module for a 180 kW Semikron Insulated Gate Bipolar Transistor (IGBT) module using 100 °C 50/50 PGW mixture at a pressure drop of 40 psi, providing a heat flux of 165 W/cm^2. In a laboratory setting, Yang et al. (*48*), with a multi-nozzle spray cooler using water as the working fluid, reached a heat flux of ~1000W/cm^2.

2.2 Jet-impingement Cooling

Jet impingement can be implemented in three basic forms: free jet (liquid jet in a vapor or gaseous ambient), submerged jet (liquid jet in a liquid ambient), and confined jet (liquid jet confined between the nozzle and heat source). From figure 5, the primary components of a free jet jet-impingement cooler can be seen. With sufficient driving pressure, a continuous jet of fluid is forced through a jet nozzle that impinges upon the heated surface. With adequate heating, bubbles will form and the jet-impingement cooler will operate as a two-phase cooler.

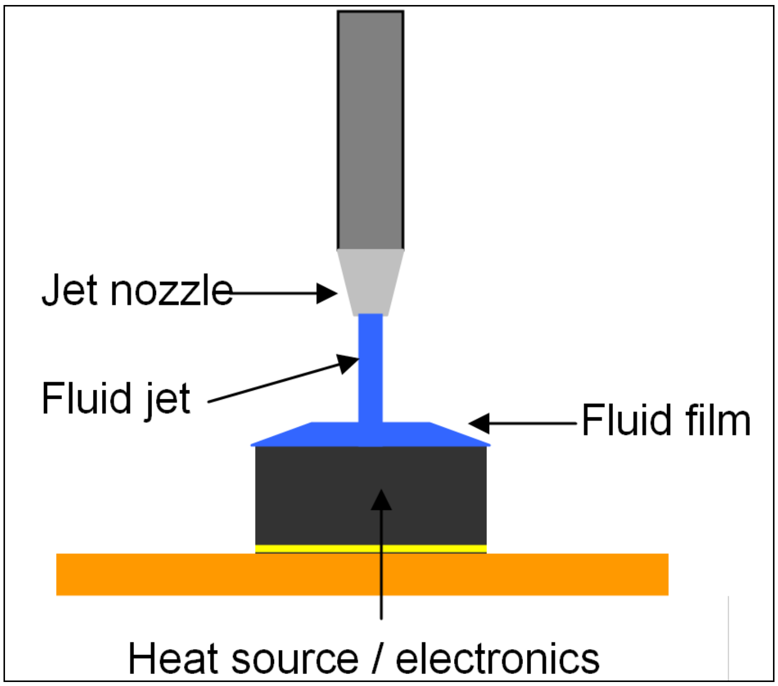

Figure 5. Primary components of a jet-impingement cooler.

The fluid flow will take two forms—an impingement zone, where the fluid is colliding with the hot surface, and a flow region, where the liquid is flowing from the impingement zone to adjacent surfaces. The multiple fluid regions (impingement and flow) give rise to the jet profile shown in figure 4, with the impingement zone having a much larger heat transfer coefficient than the flow region. Furthermore, figure 4 demonstrates jet-impingement's potential to provide significantly improved localized heat transfer coefficients compared to spray cooling. Since jets concentrate most of the cooling potential in the impingement zone, multiple jets are commonly used to create several impingement zones to cool larger areas (*28, 48, 50*). Because the fluid is continuous, unlike droplets in spray cooling, the fluid dynamics are a bit simpler. Modeling

using current Computational Fluid Dynamics (CFD) approaches is possible, but still nontrivial, and requires user-defined functions (*51, 52*).

However, the concentration of the heat transfer coefficient and associated heat removal within the impingement zone can cause significant temperature gradients within the device. This temperature non-uniformity can cause thermal stress in the electronics and potential failure. Jet-impingement is also prone to separation of the wall jet with vigorous boiling, leading to dry-out and eventual physical burn-out of the electronics (*53*). Furthermore, jet-impingement is considered an aggressive form of cooling due to the large impact momentum imparted on the heat source; the impact force may be enough to damage delicate devices requiring additional system implementation considerations (*28*).

Two-phase jet-impingement system designs, despite known shortcomings, have also demonstrated the ability to cool high-heat-flux electronic modules. A preliminary test by Copeland (*49*) demonstrated heat fluxes as high as 369 W/cm^2 in a two-phase pin fin array, using multiple slot nozzle impingement with FC-72 fluorocarbon liquid and moderate flowrates ranging from 0.075 to 0.6 l/min. Phase-change jet-impingement cooling was also demonstrated by Mudawar (*28*) on the SEM-E BTPFL-C3 avionics Clamshell Module. By using direct two-phase jet-impingement and FC-72 dielectric fluid, the BTPFL-C3 performance exceeded 10 kW, and heat fluxes as high as 255 W/cm^2 were demonstrated. Bhunia et al. (*25*) demonstrated power densities up to ~1000 W/cm^2 in a laboratory setting tailored towards high power density electronic cooling.

2.3 Microchannel Cooling

Shown in figure 6 are the primary components of a microchannel cooler. Electronic devices are typically attached to the topside of the microchannel cooler, with the backside left open for microchannel fabrication. With appropriate driving pressure, the fluid is forced through large surface area microchannels, and heat is transferred from the topside electronics, through the microchannels, and into the fluid. Similar to spray-cooling and jet-impingement, microchannel systems were initially designed as single-phase systems, but with sufficient heating the system will operate as a two-phase cooler.

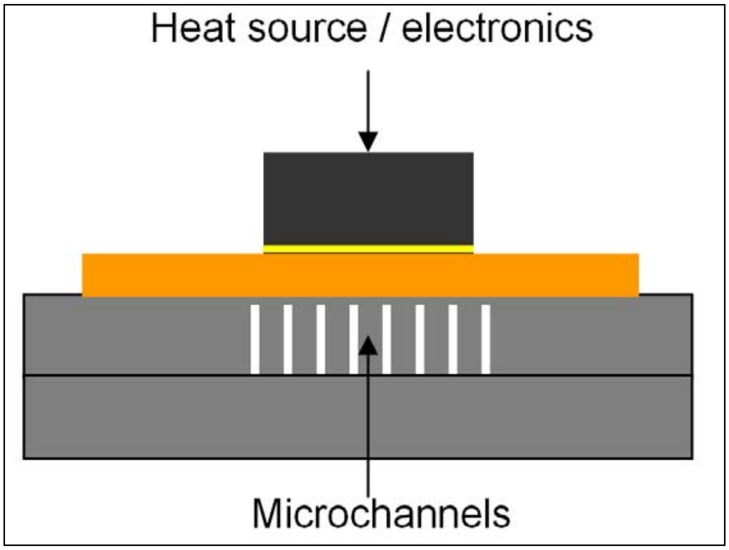

Figure 6. Primary components of a microchannel cooler.

Two-phase microchannel systems have attracted much attention as a method for high-heat flux electronic cooling. While it has been shown that flow through large surface area microchannels boosts the heat transfer coefficient in single-phase flow (*16, 54*), additional gains in the heat transfer coefficient can be obtained by allowing the liquid to vaporize along the walls of the microchannel. These systems also possess attributes like compactness and minimal coolant usage that make them attractive for electronics cooling applications.

However, two-phase microchannel heat sinks have demonstrated shortcomings that will necessitate extensive engineering knowledge to overcome in order to develop effective cooling systems. Two-phase microchannel heat sinks have significant steady-state and transient flow and pressure uncertainties, making them prone to dry-out and critical heat flux overshoot that can cause overheating and electronic burn-out. Boiling incipience and conjugate effects, such as nonuniform fin temperature, are also contributing factors to the uncertainty of two-phase microchannel cooling, limiting the practical application of such systems (*55, 56*).

Despite these impediments, two-phase microchannel designs have demonstrated the potential to provide the cooling necessary for high heat flux electronic systems. One example is a study performed by Lee and Mudawar (*13*) with R134A and HFE1700 direct and indirect microchannel cooling systems; the study reports heat fluxes as high as 840 W/cm^2 without overheating or electronic burn-out. As an extreme, a study by Mudawar et al. at Purdue University (*56, 57*) have confirmed ultra-high heat flux values as large as 27,600 W/cm^2 with subcooled water flow at high velocity in small diameter micro-tubes (D<1mm). While this performance may not be feasible with Army vehicular flowrate and pressure requirements, this study demonstrates the capability of two-phase cooling and reiterates the potential for such cooling schemes.

2.4 Technology Comparison

The three previously described two-phase cooling technologies have demonstrated the capability of providing high levels of cooling using elevated temperature coolants. The successful use of any of these techniques would potentially enable high heat flux electronics to operate on an elevated temperature coolant loop. Common to all technologies are operational uncertainty, the risk of instability under steady or transient loads, and the presence of a sudden, critical failure point. This poor understanding leads to increased margins of safety, forcing operation of the cooling component far from its peak performance, and reducing the net benefit of two-phase cooling schemes from the start.

Specific to each technique, however, are particular drawbacks that might make Army vehicle implantation problematic. With respect to spray cooling, the high driving pressure is the primary stumbling block to system compatibility. While two-phase cooling techniques all have tended to exhibit higher pressure drops than single-phase equivalents, a large driving pressure is the fundamental mechanism to spray formation. Because jets and channels are continuum flows, they generally can be operated over a wide range of flowrates, and the system compatibility challenge becomes improving performance at lower pressures. A spray component would require system level redesign to support the pressure requirement necessary for adoption into the previously mentioned cooling loop architecture.

Both the jets and the microchannels seem capable of operating over a similar pressure range. The jets have the potential for high convection coefficients, but this appears to be highly localized to the impingement region of the jet. While jet arrays can reduce this effect to some degree, an array configuration introduces additional complexity in terms of fluid routing and removal. Proper manifolding of such a jet configuration to manage fluid delivery and removal could be a nontrivial design and assembly problem (*58*). While there have been numerous single-phase microchannel studies examining complex manifolding techniques (*59*), their general purpose is to reduce the temperature rise effect of sensible heating, which two-phase approaches already mitigate via latent heat effects. Thus, standard microchannel designs with relatively simple fluid paths could provide appropriate cooling performance within the confines of our system limits.

It is also worth mentioning that the microchannel configuration can be somewhat simpler from an academic sense, with fewer geometric variables and little interaction between individual elements, making it a more tractable problem. With these factors in mind, two-phase microchannel coolers represent the most promising option for high heat flux power electronics cooling goals in Army vehicle applications.

3. Two-phase Microchannel Flow Obstacles

In general, two-phase flows are complex physical processes dependent on a variety of variables: channel shape, flow direction, fluid compressibility, interfacial tension forces, fluid wetting characteristics, momentum, pressure, temperature, fluid velocity, and channel orientation. These can all contribute toward various liquid-vapor two-phase flow patterns and regimes (*60*). As demonstrated by extensive literature (*61–78*), the mechanisms controlling two-phase pressure drops, heat transfer coefficients, and critical heat flux are inherently related to flow regimes, thus making flow regime prediction a key component of research and two-phase heat transfer design. In addition to steady-state flow concerns, the current understatement of two-phase transient behavior is also presenting a hindrance, limiting practical system implementation. If dynamic startup and transient conditions are left unmitigated in such systems, not only will system stability decrease, but the potential for system failure will increase (*79, 80*). The following sections are aimed towards describing the nature of these problems.

3.1 Microchannel Two-phase Steady-state Flow Regimes

An extensive knowledge base surrounding steady-state two-phase flow exists and, while it is hard to predict the stability with specific design parameters in mind, it typically follows the progression outlined in the following section. Figures 7a and 7b show the flow regimes for two-phase microchannel flow in horizontal and vertical orientations, respectively. With a subcooled inlet, where the temperature of the entering fluid is lower than the saturation temperature, single-phase forced liquid flow continues for a finite length until sensible heat absorption raises the temperature enough for subcooled boiling to commence along the wall. The point between liquid-forced convection and subcooled boiling is known as incipent boiling and indicates the condition where boiling can begin. There are two regions in subcooled boiling—the first where the core liquid is very subcooled and boiling only occurs on the walls, and the second where the fluid has a higher temperature, but still below the saturation temperature, and vapor bubbles can be found in the core of the microchannel. This difference occurs because the bubbles immediately condense in the liquid core of very subcooled liquid; a liquid core closer to the saturation temperature, however, allows bubbles to persist in the bulk flow without condensing. The interface between these two subcooled boiling regimes is the point of net vapor generation. The next bulk region is the saturated boiling regime, which consists of bubbly flow, slug (plug) flow, and annular flow. In the saturated boiling region, the vapor quality can range from 0 (0% vapor and 100% liquid by mass) to approximately 1 (100% vapor and 0% liquid).

3.2 Thermal Failure and Critical Heat Flux (CHF)

From a design perspective, there are thermal implications associated with the flow regimes discussed above. As explained by Incropera et al. (*21*), and indicated in figure 7a, the transition

from subcooled boiling, to bubbly flow, slug flow, and annular flow represents increases in vapor quality, heat transfer coefficient, and thermal performance. The annular flow region persists until wall dry-out and, eventually, vapor forced convection occur. This transition, and subsequent dry-out, results in severely reduced heat transfer coefficients due to the much lower convective performance of vapor compared to liquid. Because the system will be able to dissipate less heat after a reduced convection coefficient due to dry-out, temperatures in those regions can be much higher and result in system overheating and burn-out.

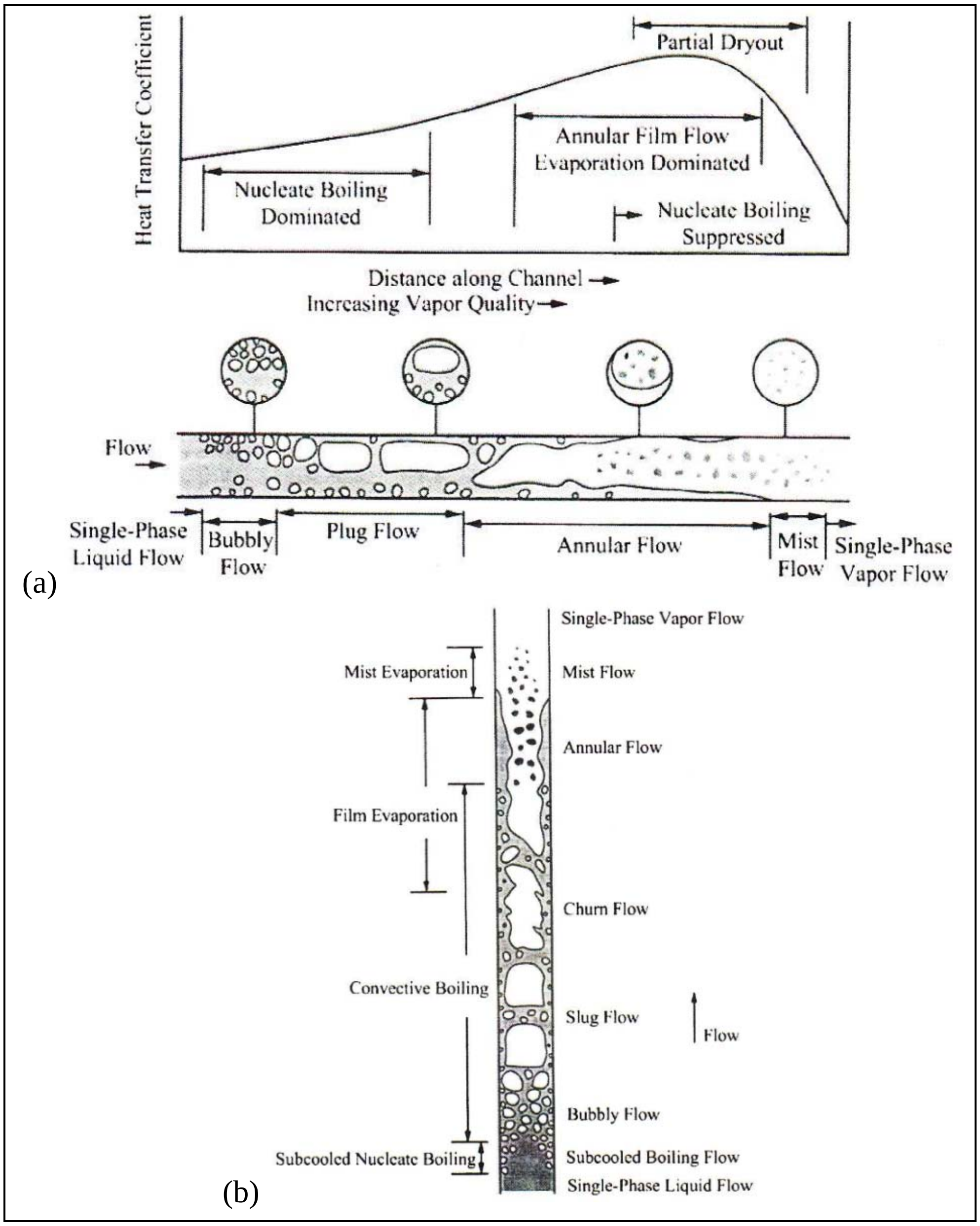

Figure 7. Ideal microchannel 2-phase boiling flow regimes in (a) horizontal and (b) vertical orientations; from (*60*).

As discussed previously and shown in figure 8, there is a point within the microchannel system where dry-out and possibly critical thermal failure can occur. This condition will be dependent on the amount of heat being added to the system, and the thermal limit inducing this condition has been described as the critical heat flux (CHF) or boiling crisis. Referring to figure 8, Point A indicates the transition from single-phase to boiling heat transfer. As expected with the latent absorption and improved convection due to boiling, the slope of the curve increases greatly to the right of Point A and higher heat flux is obtainable with minimal wall temperature rise. Point C indicates the location where excess heating begins to causes partial or total dry-out, and a sudden decrease in heat transfer coefficient is observed. This location typically occurs in the saturated annular flow regime, but it can also be seen in subcooled boiling, where excess heat causes a vapor blanket to form on the walls of the channel even though the actual vapor quality for the entire channel is still relatively low. While operating the system close to CHF typically yields annular flow and higher heat transfer coefficients, exceeding the CHF limit would cause overheating of the channels and potential electronic failure. In general, the design objective in creating a two-phase microchannel cooler is to induce boiling at the earliest point in the microchannel without encountering dry-out at the end of the channel—or, in other words, without exceeding the critical heat flux for that flow. However, the inability to predict flow regimes and subsequent CHF design values has required a factor of safety be used in the design of any of these cooling systems, and operating too close to any perceived non-recoverable failure point is unacceptable. As shown on figure 8, a factor of safety would move system operation down on the curve and provide less than optimized heat flux. The power derating that would currently be needed to guarantee the system stay below these estimated critical failure points could likely render the cooling solution less effective than current single-phase options. Research must be directed at higher fidelity prediction of these failure limits and better design for stable operation.

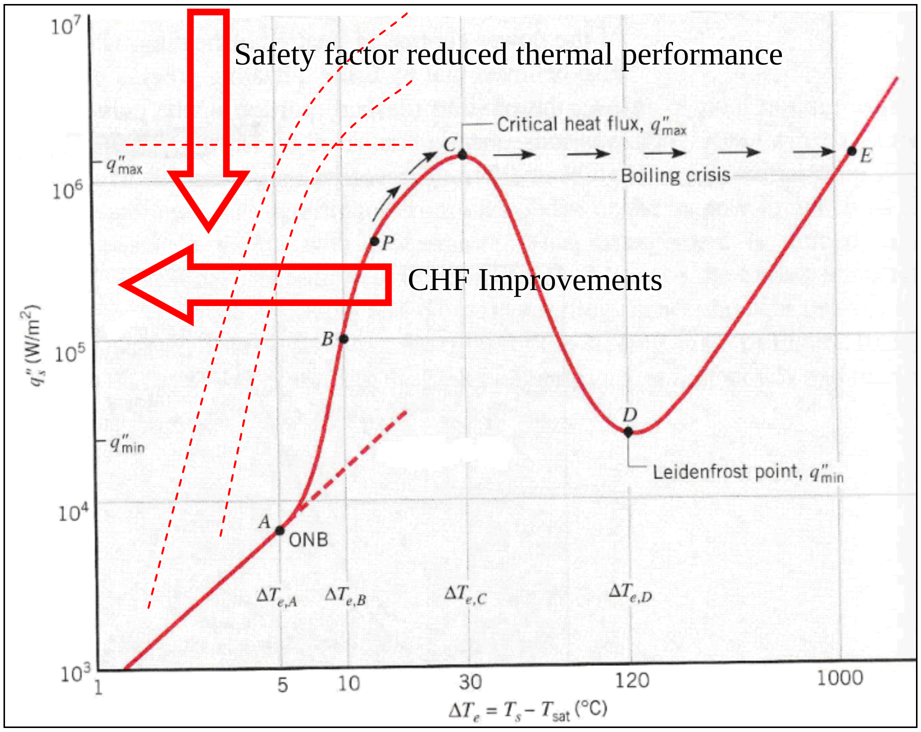

Figure 8. Critical Heat Flux for Water @ 1atm. Point A is the onset of boiling and graph follows typical flow progression up to annular flow and CHF at Point C; from (*21*).

It is worth noting that since microchannels are typically cut into materials with moderate to large thermal conductivities, local critical heat flux may not always cause failure of the entire cooling system. From a practical perspective, vapor formation and dry-out of a portion of the microchannel cooler will cause local overheating. Due to a large thermal conductivity, however, the heat can be redistributed to other other areas of the microchannel heat sink that can accommodate the heat dissipation. Therefore, the reliability and robustness of a two-phase microchannel cold plate can be increased by using materials with high thermal conductivities.

In microchannel applications where the fluid is entering in a liquid state, it is important to determine if there are any barriers to vapor formation; vapor formation indicates the onset of boiling and is necessary in a two-phase cooling system to achieve the thermal benefits associated with boiling. It has been presented in the literature (*28*, *81*, *82*) that vapor formation and boiling inception is a key factor to system performance and critical heat flux increases. By creating surface imperfections or nucleation sites, boiling enhancement in the form of micro-porous surfaces, painted diamond particles, surface roughening, and painted silver flakes have demonstrated boiling inception temperatures 75–85% lower than smooth surfaces with system CHFs as much as 224% higher (*82*–*84*); this would indicate a shift up and left on the curve in figure 8. Also in figure 8, improvements in CHF represent lower wall superheat necessary for boiling inception and higher allowable heat flux before potential failure occurs. As an avenue for cooler performance enhancement, engineers have explored combining microchannels with surface modification techniques previously discussed for further improvement in boiling

inception and critical heat flux. It is important to note that while smaller microchannels and rougher surfaces improve the thermal transport qualities, they also increase the pressure drop across the device (*85, 86*). For Army applications, significant design attention would be necessary to balance CHF improvements and associated pressure drops.

3.3 Channel Orientation

There has been little experimental attention paid to inclined and declined microchannel flows, despite obvious technical importance; many Army vehicles can be expected to climb significant grades, therefore, the associated electronics and coolers would encounter these inclines as well. As described by Hewitt (*63*), inclined and declined two-phase microchannel flows can be expected to demonstrate qualities of horizontal and vertical orientations combined.

Due to buoyancy effects, gravity can shift the distribution of the different fluid phases in the channel. Generally, horizontal microchannel flows demonstrate asymmetry due to gravity, causing stratification with the liquid at the bottom of the channel and vapor to occupy the top; this is illustrated in figure 9a, which is in contrast to the more uniform liquid-vapor distribution shown for a vertical configuration in figure 9b. This asymmetry and stratification causes thermal complications—possible intermittent drying and rewetting of the upper surface of the tube over short distances, or total dry-out of the upper surface in long channels that can cause overheating and burn-out of the channel and electronics (*87–91*). This scenario is exacerbated by potential non-uniform conductive heat flow around the perimeter of a microchannel, causing variation in the boiling regime and associated heat transfer coefficient within the channel (*55, 92, 93*). To mitigate these effects, studies have shown that higher inlet liquid velocities and high thermal conductivity microchannel materials lead to more symmetric flow and reduced non-uniform heat flow (*60*).

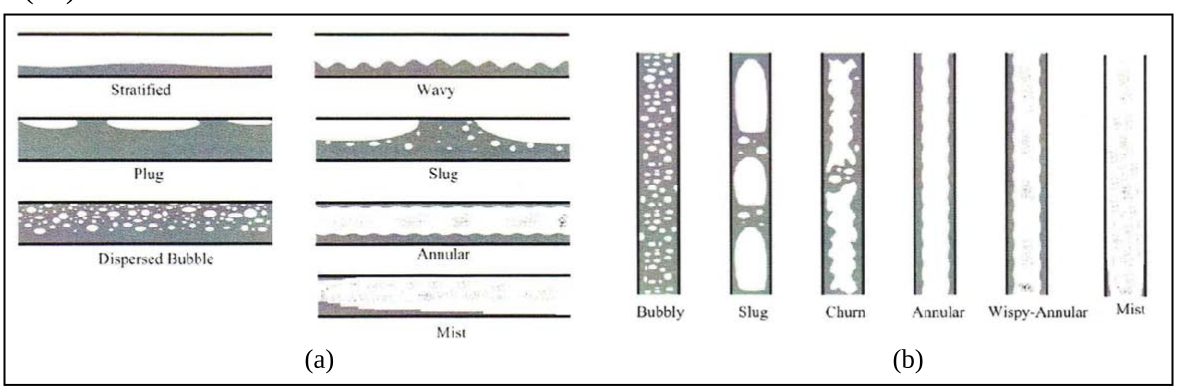

Figure 9. Non-Ideal microchannel 2-phase boiling flow regimes in (a) horizontal and (b) vertical orientations; from (*60*).

3.4 Microchannel Two-phase Transient Effects

As discussed previously, significant attention has been paid to steady state two-phase flow regimes. However, startup and dynamic conditions in a two-phase cooling system are very complex transient phenomena that need to be considered. Specifically, special attention must be

paid to variations in heat load, mass flowrate, electronic component temperature, and coolant fluid temperature. Two studies exploring the transient and startup of a novel pumped two-phase cooling loop were presented (*79, 80*), demonstrating acceptable performance in the form of stable temperature, pressure, and flow. While there has been some experimentation concerning two-phase transients, there is an overall lack of documented experimental results for two-phase microchannel transient performance. The result is systems designed with larger safety margins, reducing the net benefit of two-phase cooling schemes.

3.5 Microchannel Flow Summary

In general, several statements can be made about the status and future path of two-phase microchannel cooling techniques. While there are a significant number of studies in microchannels, there is no proven consistency for the same fluids under the same test conditions. This may be in large part due to the inconsistent and subjective visual-only methods used to characterize flow regimes. While several review papers outlining prediction of two-phase microchannel heat transfer characteristics and flow maps are available (*60, 94*), widely accepted theoretically based flow regime flow maps have not been established (*60, 95*). Research must be directed in several directions to provide systematic knowledge and promote higher fidelity prediction of two-phase flow in microchannels:

1. Effects of channel shapes, channel orientation, fluid properties, and other physical variables.
2. Steady-state flow regimes and associated failure limits.
3. Transient conditions to better design for stable and dynamic operation.

4. Two-phase Macro/Microchannel Commercial Success

Two-phase macrochannel coolers are being implemented as an intermediary step between conventional single-phase methods and future two-phase microchannel designs. At the cost of reduced thermal performance, macrochannel designs have a more extensive research basis and are currently considered less complex and easier to implement than their higher-performing microchannel counterparts. With the design philosophy of "simplicity and effectiveness" in mind, Parker Hannifin has developed a high-voltage power system for military vehicle applications using a two-phase macrochannel cooling technology (*14*). They are currently past the research and development stages with their technology and are deep into commercialization and shipping.

Figure 10 is a photograph of a Parker Hannifin 450A IGBT two-phase cooling module. Their system uses R134A dielectric fluid as the coolant, a low flowrate positive displacement pump (not shown), specially designed two-phase corrugated copper cold plates, and a vapor condenser

(not shown). R134A was chosen because it is a well-characterized refrigerant with widely commercially available pumps and components; as previous fluid selection information implies, other fluids may be used with appropriate pump and component modifications.

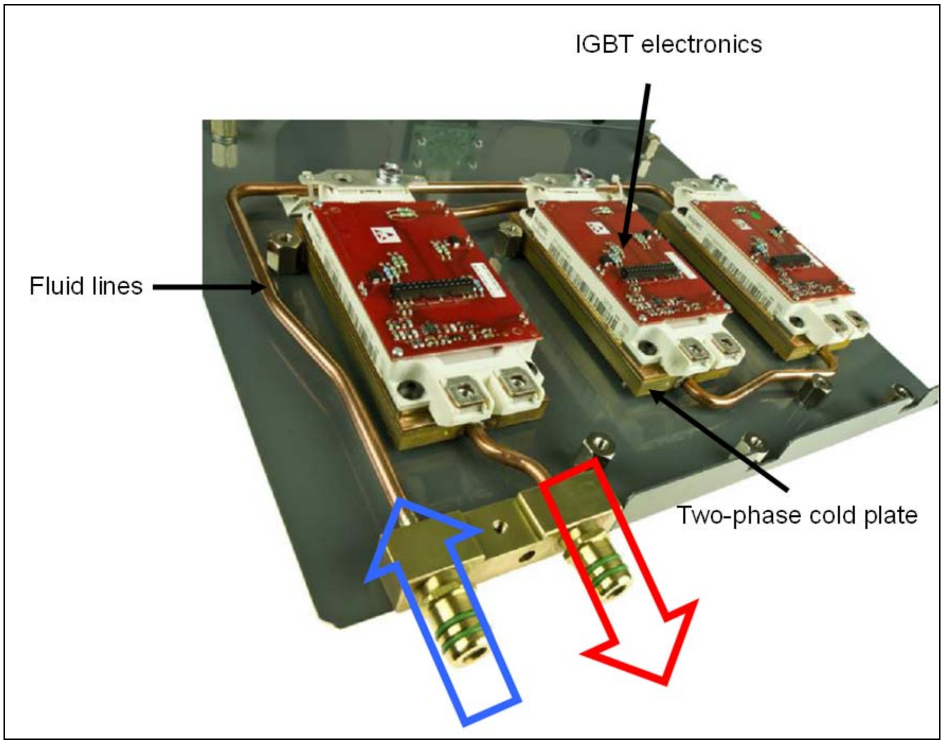

Figure 10. Two-phase Parker Hannifin 450A IGBT cooling module; from (*14*).

The performance of this module is characterized by a power dissipation comparison to other cooling technologies. As outlined in table 1, five different cooling schemes were employed to observe the relative performance of the two-phase cooler.

Table 1. Parker-Hannifin inverter test cases; from (*14*).

Case	*Thermal Solution*	*Description*	*Fluid Flow Rate per Cold Plate LPM (GPM)*
A	Forced air cooled extruded heat sink	Aluminum, 14:1 fin ratio, 150CFM airflow	N/A
B	Water-cooled liquid cold plate, standard	Press-fit copper tubing in aluminum body	7.57 (2.0)
C	Water-cooled liquid cold plate, custom	D-shape copper tubing circuit aligned to die locations	7.57 (2.0)
D	Water-cooled liquid cold plate, custom	Convoluted Al fin pack, vacuum brazed machined Al surface	7.57 (2.0)
E1	VDF R-134a pumped liquid cold plate (450A IGBT)	Convoluted Cu fin pack, brazed, machined Cu surface	1.51 (0.4)

Figure 11 shows that the two-phase cooling technique (VDF Copper, vaporizable dielectric fluid) is superior in cooling capacity to all other tested single-phase cooling methods. Not only does the VDF Copper two-phase cooling method have a significantly reduced flowrate (compare 0.4 to 2 GPM in table 1), the cooling capacity is about 40% better than the next highest performing single-phase cooling technique (*14*).

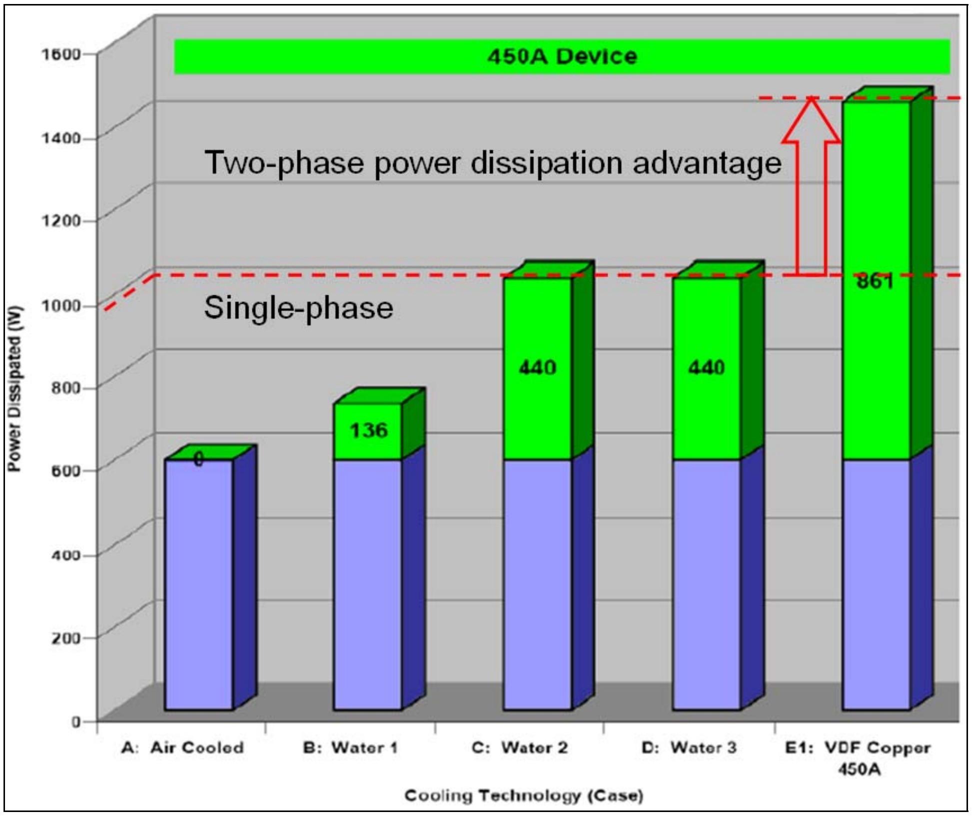

Figure 11. Cooling Performance of the two-phase Parker Hannifin cooler compared to single-phase cooling techniques; from (*14*).

Pumping power consumption, an advantage mentioned earlier, is a major benefit when considering a two-phase cooling system. The pumping power necessary to cool a 1 kW heat load with the R134A two-phase copper heat sink was measured to be 12 W. With an equivalent load and similar temperature limits, a single-phase water aluminum pressure-fitted cold plate required 295 W of pumping power to match the cooling potential of the two-phase method. This system also has the added benefit of an isothermal heat sink temperature, which reduces the thermal stress on silicon chips and improves reliability (*14*).

Compared to standard single-phase cooling techniques, the above commercial example demonstrated the potential for isothermal cold plates, pumping power reduction, flowrate reduction, and improved heat dissipation. Despite these achievements, areas of improvement relating to CHF need to be explored. As previously indicated, approaching CHF in annular flow with high vapor qualities would be beneficial to increase heat flux, reduce flowrate, and minimize size and weight of associated components. However, this module was designed around selecting an appropriate vapor quality maximum value (70%) as not to exceed CHF and prevent dry-out from occurring within the cold plate (*14*). Aforementioned steady-state uncertainties and transient conditions played a large role in this module’s cautious design, and improved research attention in these areas should be a major focal point to further increase predictability and performance of two-phase systems.

5. Conclusions and Recommendations

The difficulties in designing a hybrid electric propulsion system for Army vehicular applications have consistently called for advanced thermal management technologies. However, single-phase forced-convection liquid cooling has reached a mature state in the academic and commercial worlds, and it is highly unlikely that the order-of-magnitude heat flux improvements desired for future systems will be obtainable by this technology under the current set of platform constraints. The key limitations can be summarized as:

1. Minimum allowable orifice size—particles and sediment in the primary engine loop limit cooling scheme geometries and may cause clogging.
2. Fluid limitations—the primary cooling loop has been designed for an EGW coolant, which is not an ideal fluid for use in electronics cooling.
3. Pumping inflexibility—fluid delivery requirements can not be optimized for the electronic cooling components due to engine coolant flow and pressure specifications.
4. Single-phase fluid limitation—because the system is not designed to accommodate mixed gas/liquid flow, there is currently little opportunity to implement two-phase flow.

This paper has examined electronics cooling on the Army vehicle platform, with particular focus on the factors limiting the use of two-phase microchannel cooling technology. Based on the platform specific limitations just described and the current unknowns in the academic community on two-phase microchannel cooling, the following research directions are recommended for achieving the thermal performance necessary for future systems:

1. Determine effects of EGW additives—additives may significantly influence two-phase cooling performance and behavior compared to “pure” EGW.
2. Steady-state two-phase flow risk reduction—reliable and accurate models of flow stability and CHF must be established to enable adoption in real systems without significant performance-reducing safety margins.
3. Transient two-phase flow risk reduction—the effects of two-phase cooling under dynamic conditions must be understood to improve system reliability.
4. Study vehicle architectures compatible with two-phase cooling—examining the system-level costs and benefits of multi-loop architectures would help identify potential Army vehicle cooling solutions.

Single-phase cooling techniques do not possess the cooling capability necessary to enable future Army power-dense hybrid-electric vehicle applications, especially with the requirements for

elevated coolant loop temperatures. Despite demonstrated performance, two-phase cooling has significant technical uncertainties, causing a level of implementation risk. Only by reducing the risk of two-phase cooling solutions while maintaining the promised power density improvements will system builders be willing to consider the technology for implementation in future platforms.

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Appendix A. Typical Coolant Fluids and Associated Properties

Cooling Fluid Description			Specific Heat@25C	Latent Heat@25C	Density@25C	Boiling Point	Flammable	Ignition Temp.	ODP	GWP	NFPA		HMIS	
	Company	Product Name	kj/kg/K	kj/kg	kg/m³	°C	Yes/No	°C	R11=1	100yr	health	flammability	health	flammability
De-Ionized Water		H2O	4.13	2440	1002	100	No	N/A	0	<1	0	0	0	0
Ethylene glycol/water 50/50	various	EGW	3.58	865	1088	106	No	N/A						
Propylene glycol/water 50/50	various	PGW	3.55	980	1041	106	No	N/A						
1,1,1,2 tetra fluorethane	various	HFC-134a	1.4	178	1210	-26	No	750	0	1300	1	0	1	0
1,1-dichloro-1-fluroethane	various	HCFC-141b	1.15	226	1234	32	Yes	325	0.086	700	1	0	1	0
1-chloro-1,1-difluoroethane	various	HCFC-142b	1.3	217.8	1.12	-10	Yes	632	0.043	2300	1	0	1	0
Dichlorodifluoro methane	various	CFC-12	1	139	1311	-29.8	No	>750	0.82	10600	2	4	2	4
Chlorodifluoro methane	various	HCFC-22	1.26	182.7	1191	-40.8	No	N/A	0.034	1700	2	1	2	1
perfluorocarbon	3M	FC-87	1.1	103	1650	30	No	N/A	0	8900	3	0	0	0
perfluorocarbon	3M	FC-72	1.1	88	1700	56	No	N/A	0	9000	3	0	1	0
perfluorocarbon	3M	FC-77	1.1	89	1780	97	No	N/A	0	9000	3	0	1	0
hydrofluoroether	3M	HFE-7000	1.3	142	1400	34	No	415	0	400	3	1	0	1
hydrofluoroether	3M	HFE-7100	1.17	112	1520	61	No	405	0	320	3	1	0	1
hydrofluoroether	3M	HFE-7200	1.22	119	1420	76	No	375	0	55	3	0	1	0
methylsiloxane	Dow Corning	Cleaning Agent 1	1.72	194	850	100	Yes	341.1	0	<10	1	3	N/A	N/A
methylsiloxane 60% w/alcohol	Dow Corning	Cleaning Agent 2		255	770	98	Yes	365	0	<10	1	3	N/A	N/A
butane/isobutane/propane blend	Duracool	Duracool 12a	2.56	343.5	526	-31.5	Yes	891	0	~20	1	4	1	4
propane/butane blend	Enviro-safe	ES-12	2.574	354.2	530	-30.4	Yes	863	0	~20	1	4	1	4
HFC-134a/HCFC-142b blend	Technical Chemical	Freeze 12	1.4	180.7	1189	-25	No	>150	0.01	1500	2	2	2	2

List of Symbols, Abbreviations, and Acronyms

ARL	Army Research Laboratory
CFD	Computational Fluid Dynamics
CHF	Critical Heat Flux
DoD	Department of Defense
DOE	Department of Energy
EGW	Ethylene Glycol and Water
IGBT	Insulated Gate Bipolar Transistor
NASA	National Aeronautics and Space administration
PGW	Propylene Glycol and Water
Sodium MBT	Sodium Mercaptobenzothiazole
SWAP	Size, Weight, and Power
TARDEC	Tank Automotive Research, Design, and Engineering Center
VDF	Vaporizable Dielectric Fluid

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